Thermodynamic investigation of asynchronous open inverse air cycle integrated with compressed air energy storage

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Adriano MILAZZO[(a)], Niccolo GIANNETTI[(b)], Seiichi YAMAGUCHI[(c)], Kiyoshi SAITO[(c)]

[(a)] University of Florence, Department of Industrial Engineering,
Via di Santa Marta 3, Florence 50139, Italy, adriano.milazzo@unifi.it
[(b)] Waseda Institute for Advanced Study, Waseda University
1-6-1 Nishiwaseda, Shinjuku-ku, Tokyo, 169-8050, Japan, niccolo@aoni.waseda.jp
[(c)] Waseda University, Department of Applied Mechanics and Aerospace Engineering
3-4-1 Okubo, Shinjuku-ku, Tokyo, 169-8555, Japan, sei_yamaguchi@aoni.waseda.jp

ABSTRACT

An integrated system for heating, cooling and compressed air energy storage (CAES) is analyzed from a thermodynamic point of view. The system is based on asynchronous air compression and expansion, in order to take advantage from the daily ambient temperature oscillations and energy cost variations. The analysis is intentionally kept on a fundamental level, without explicit reference to specific components, in order to enlarge the choice of potential applications. Effects of losses in compressor, expander and heat exchangers, as well as heat transfer in the CAES, are included. The proposed system, once optimized and experimentally validated, could become viable options in the wide arena of demand-side energy management.

Keywords: Air-refrigerant, CAES Energy Storage, Thermodynamic Investigation, Asynchronous.

1. INTRODUCTION

Among energy storage systems, Compressed Air Energy Storage (CAES) has been often proposed as an environmentally and technically feasible option. The first CAES plant in Huntdorf, built in 1978, was aimed to conciliate the variable grid energy demand with the constant electric energy production from nuclear or coal, big size power plants (Crotogino et al., 2001). Air compression during the energy storage phase is basically adiabatic, so that the mechanical power used for air compression is equal to its enthalpy variation:

$$\dot{W} = \dot{m}c_P(T_C - T_{amb}) \qquad \text{Eq. (1)}$$

Temperature at the end of compression may be computed once pressure ratio and compressor efficiency are known. An ideal compressor would give the minimum exit temperature, i.e.:

$$T_C = T_{amb}\left(\frac{P_C}{P_{amb}}\right)^{\frac{k-1}{k}} \qquad \text{Eq. (2)}$$

where $k = c_P/c_V$ is 1.4 for ambient air. Even for said ideal compressor, when the design pressure of 70 bar adopted at Huntdorf and an ambient temperature of 20°C are considered, the exit temperature would be around 714°C. When a likely compressor efficiency is accounted for, this value would increase by a further 10%. The significant amount of thermal energy contained in the compressed air (basically equal to the work paid for compression) is wasted to the cavern where the compressed air is stored. Therefore, before expansion, air must be heated up. If not so, expanding from e.g. 20°C across the same pressure ratio (Eq. 3), air would reach a temperature as low as -186°C (actually a bit higher when considering energy loss within the turbine).

$$T_E = T_{amb}\left(\frac{P_{amb}}{P_C}\right)^{\frac{k-1}{k}} \qquad \text{Eq. (3)}$$

Obviously this would cause severe technical problems. This simple calculation shows two things:

- A CAES designed without any account for the management of the huge amounts of thermal energy at play is very inefficient.
- A side product of any CAES plant may be heating and cooling for useful purposes.

The first point has been addressed by many authors, giving rise to the so-called CAES-TES concept, i.e. integration of CAES with a Thermal Energy Storage. A European project has been funded under EC DGXII contract ENK6 CT-2002-00611 and has led to the design of the so-called AA-CAES, ("Advanced Adiabatic CAES") (Bullough et al., 2004). Proponents of the AA-CAES concept insisted for the need to raise the TES temperature, in order to increase the exergy of the stored thermal energy, even if this causes severe technical problems. Grazzini and Milazzo (2012) have shown that this idea is not productive, the energy recovery efficiency being directly proportional to the number of compression/expansion stages and hence inversely proportional to the TES temperature (actually the most efficient CAES should have an isothermal compression/expansion and use the environment as TES).

The integration of CAES with cooling/heating has been proposed, among others, by A. Facci et al. (2015). The use of air as a working fluid in a refrigeration cycle has some unique advantages:

- Air is an absolutely "natural" working fluid, perfectly safe for users and environment.
- Air is available everywhere at zero cost, so that the working cycle may be opened towards the environment, eliminating a heat exchanger.
- Air is used by humans and animals for breathing and hence may combine the functions of heat subtraction/delivery and removal of indoor-generated pollutants.

Normally the cooling systems using air as a refrigerant use an inverse Brayton cycle, which may have two main configurations:

- "low pressure" cycle: the high temperature heat exchanger is eliminated and the expansion of atmospheric air (possibly the exhaust air from the cool space) is used to produce a low-temperature and low-pressure flow that feeds the low-temperature heat exchanger. The low-pressure air will subsequently be compressed and exhausted at high temperature to the environment.
- "high pressure" cycle: atmospheric air is compressed, cooled in a high temperature heat exchanger and then expanded back to atmospheric pressure, reaching low temperature. This low temperature air may be directly introduced in the cool space.

Grazzini and Milazzo (2010) have shown that the "low pressure" cycle has invariably higher efficiency. However, in the case of low-temperature refrigeration (e.g. fast freezing of food), the chance of avoiding frost formation upon the cold heat exchanger suggests to prefer the "high pressure" cycle. In this case, an efficient regenerator may greatly improve the cycle performance (Giannetti and Milazzo, 2014).

The combination of these ideas may produce several different schemes and satisfy various requirements. Air compression/expansion is used when temperature increase/decrease is needed. Compressed air is used to store energy, but given that quite big volumes are needed, the CAES surface may be used also as a heat exchanger, as long as enough time is available to reach thermal equilibrium with the surroundings. Energy storage may be used to decouple in time the cooling request and the work consumption. Besides, daily variations in the ambient temperature may be used in order to optimize the system efficiency.

2. BASIC ASYNCHRONOUS CONFIGURATION

A first, very simple scheme following the above-descripted principles is shown in Fig. 1. The main components are a compressor "C", a heat exchanger, a storage "CAES" and an expander "E". This configuration allows to perform the basic services required, i.e. energy storage during low-load hours (if good tariffs are available e.g. at night), production of hot water (e.g. for sanitary use), production of electric energy during high-load hours and production of cold air for cooling (e.g. air conditioning). Typically, compressed air will cool down in the CAES and some humidity will condense. Therefore, if condensed water is drained from the CAES, de-humidification will be performed as well. However, for now, air humidity has been neglected.

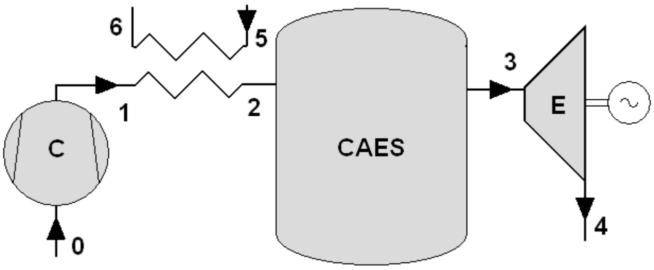

Figure 1: Base configuration

Note that if air compression is performed at night, its temperature will be lower and hence the work required will reduce. Produced electricity may be consumed on-site or sold, if possible. The compressor may be a fully commercial unit, while the expander is somewhat less common and may require a specific design. Both units need to be completely oil free, in order to avoid contamination of breathable air delivered at point 4. In order to evaluate the performance of such a system, we may choose a representative environmental condition. For example, we have taken a very hot summer day in Florence (central Italy) and we have drastically simplified the temperature behaviour along the 24 hours by a sinusoidal function.

$$T_{amb}(\tau) = T_m + \Delta T cos 2\pi \left(\frac{\tau}{24} + \frac{1}{3}\right) \quad \text{Eq. (4)}$$

where τ is the time in hours. Maximum and minimum temperatures are 38°C at 4 pm and 22°C at 4 am respectively. Compression and expansion may be seen as adiabatic. Polytropic efficiency η_{pol}, which may be kept constant for any compression/expansion ratio (Grazzini and Milazzo, 2010), is preferable to isentropic efficiency. In this way, given the exponent of the ideal compression/expansion already used in equations 2 and 3, we may simply get the exponents for the compression and the expansion as in Eq. (5).

$$\lambda_C = \frac{k-1}{k}\frac{1}{\eta_{pol}} \quad ; \quad \lambda_E = \frac{k-1}{k}\,\eta_{pol} \quad \text{Eq. (5)}$$

Hence, referring to Fig. 1, $T_1 = T_0\beta^{\lambda_C}$ and $T_3 = T_4\beta^{\lambda_E}$ where β is the compression/expansion ratio. Mechanical compression / expansion power is given by Eq. (6) and Eq. (7):

$$\dot{W}_C = \dot{m}c_P T_0\left(\beta^{\lambda_C} - 1\right) \quad \text{Eq. (6)}$$

$$\dot{W}_E = \dot{m}c_P T_4\left(\beta^{\lambda_E} - 1\right) \quad \text{Eq. (7)}$$

The heat exchanger may be qualified by an effectiveness as in Eq. (8), where the heat capacity $(\dot{m}c_P)_{min}$ may pertain to water or air according to the respective flow rates. In order to minimize losses, the two heat capacities should be equal.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{m}_{air}c_P(T_1 - T_2)}{(\dot{m}c_P)_{min}(T_1 - T_5)} \quad \text{Eq. (8)}$$

A suitable model for the CAES should include mass and heat exchange. We may write mass conservation:

$$\frac{dm}{d\tau} = \dot{m}_{in} - \dot{m}_{out} \quad \text{Eq. (9)}$$

At near-ambient conditions, air closely follows ideal gas law (volume is constant):

$$\frac{dP}{P} = \frac{dm}{m} + \frac{dT}{T} \quad \text{Eq. (10)}$$

Substituting and solving:

$$\frac{dP}{d\tau} - \frac{P}{T}\frac{dT}{d\tau} = \frac{RT}{V}(\dot{m}_{in} - \dot{m}_{out}) \quad \text{Eq. (11)}$$

The conservation of specific energy e, work being absent, yields:

$$\frac{d(me)}{d\tau} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} + \dot{Q} \quad \text{Eq. (12)}$$

whence:

$$\frac{dm}{d\tau}c_V T + mc_V\frac{dT}{d\tau} = \dot{m}_{in}c_P T_{in} - \dot{m}_{out}c_P T_{out} + \dot{Q} \quad \text{Eq. (13)}$$

Recalling Eq. (9) and solving:

$$mc_V\frac{dT}{d\tau} = \dot{m}_{in}(c_P T_{in} - c_V T) - \dot{m}_{out}(c_P T_{out} - c_V T) + \dot{Q} \quad \text{Eq. (14)}$$

Heat transfer between air within the CAES and the surroundings may be calculated by a global heat exchange coefficient:

$$\dot{Q} = UA(T - T_{amb}) \quad \text{Eq. (15)}$$

Equations (11) and (14) form a system of first-order differential equations that, given suitable boundary conditions, yields the instantaneous pressure and temperature within the CAES. Just to make an example, we may fix the CAES volume to 10 m^3, a size commercially available for compressed air reservoirs operating at 12 or 16 bar. The maximum operating pressure depends on the power of the compressor. Again we try to maintain the calculation as simple and general as possible. Compressor and expander flow rates are related to compression/expansion ratio by a performance curve that depends on the type of machine (scroll, screw, centrifugal, etc.). Here we assume that a suitable control strategy will be used such that the power absorbed/delivered is constant. Therefore, equations 6 and 7 are used to find the instantaneous flow rate from the fixed power and the variable pressure ratio. In summary, basic parameters are:

Table 1. Simulation parameters for the base configuration

CAES volume	V	10 m^3
Compressor power	$\dot{W}_{comp}$	1 kW
Water inlet temperature	T_5	21 °C
Cool ambient temperature	T_7	26 °C
Polytropic efficiency for compressor and expander	η_{pol}	0.75
Heat exchanger effectiveness	ε	0.7
Compression phase	From 0 to 8 a.m.	
Expansion phase	From 8 a.m. to 24	

Results of a one-day simulation are shown in Fig. 2. Limiting the compressor power to 1 kW allows a maximum CAES pressure below 12 bar. Higher power could be achieved if the maximum operating pressure were 16 bar. Another relevant parameter is the minimum pressure within the CAES. In order to increase the performance, this pressure must be kept as close as possible to the ambient pressure (here is 1.2 bar). A prominent feature is the mutual position of the CAES and ambient temperature curves. Apart from the sharp increase at compression start and corresponding decrease at expansion end, the CAES temperature closely follows the ambient temperature, staying above during the compression phase and below in the expansion phase. Apparently the system absorbs thermal energy from the environment during the day and gives it back during the night. By simple integration, one can calculate the cooling effect along the expansion phase (Eq. 16) and the heating effect along the compression phase (Eq. 17). The results are shown in Table 2.

$$Q_{ref} = \int_{discharge} \dot{m}_{out}c_p(T_7 - T_4)d\tau \quad \text{Eq. (16)}$$

$$Q_{heat} = \int_{charge} \dot{m}_{in} c_p (T_1 - T_2) d\tau \qquad \text{Eq. (17)}$$

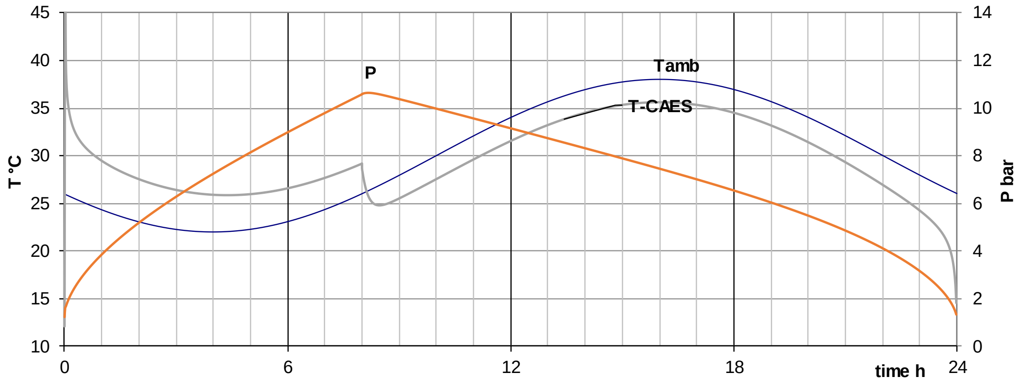

Figure 2: Simulation results for the base configuration

Table 2. Performance of the base configuration

Compression work	8 kWh
Expansion work	6.4 kWh
Net work input W	1.6 kWh
Heating effect Q_{heat}	6 kWh
Cooling effect Q_{ref}	6.1 kWh
Global COP=(Q_{ref} + Q_{heat})/W	7.56
Cooling COP=Q_{ref}/W	3.76

Closer scrutiny reveals that this configuration needs some refining. The air temperature at compressor exit reaches very high temperature and hence a staged compressor with intercooling would be necessary if the heating effect has to be transferred to water at ambient temperature. On the other hand, the air temperature at expander exit is very low, so that a staged expansion could be used, introducing an intermediate heat exchanger. Ideally these modifications could even improve the system performance, but actually side effects like pressure losses within heat exchangers, piping, etc. are likely to bring it back to the values shown in Table 2 or even below. Furthermore, this configuration has some draw-backs:

- Hot water is produced during the night, when probably there is no request. A hot water storage is therefore needed, adding complexity and losses to the system.

- The coupling of the expander with an electric generator may be difficult, if the air expansion is performed in a small turbine that must rotate at high speed, in order to reach a good efficiency.

3. ALTERNATIVE CONFIGURATION

In order to partially solve the problems mentioned in the previous section, we may conceive a second configuration as shown in Fig. 3. Electric energy production has been eliminated, as may be convenient in many situations where grid energy is available and self-generation is too difficult or scarcely remunerative. Therefore, two compressors are now provided: the upper one fills the CAES during night hours as before, while the other is meant to absorb the work delivered by the turbine. In this way, the compressor/turbine group is not mechanically linked to any electric motor and hence enjoys complete freedom in terms of rotating speed.

A suitable unit (with slight modifications if needed) could be found among turbine/compressor groups used for aviation air conditioners. These units are highly developed, compact (they rotate at high speed) and reliable (Mahindru and Mahendru, 2011). Obviously the work delivered by the turbine will not suffice to compress the same flow rate which expands through the turbine. Therefore, in daytime the CAES will be progressively emptied, consuming energy that was stored at night. A heat recovery exchanger may be introduced on the cool exhaust air from the refrigerated space, in order to pre-cool the ambient air at inlet.

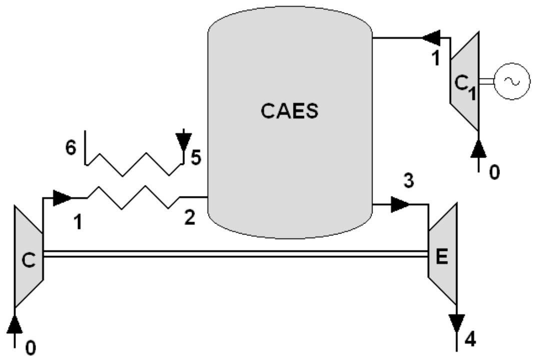

Figure 3: Alternative configuration

4. CONCLUSIONS

The proposed systems, whatever the configuration of choice, clearly need significant improvements in order to become technically feasible. Before proceeding further in the analysis, some fundamental choices should be made. First of all, one should fix a size range of interest, in order to specify the type of compression/expansion machines and hence have their performance maps. As mentioned above, in order not to contaminate the air and make it unbreathable, the set of usable technologies is restricted to oil-free compressors. An evident problem of the system is being very bulky. However, the components should be relatively simple and inexpensive. On the other hand, this system uses the safest and cheapest among natural refrigerants. The open cycle not only saves a heat exchanger, but also avoids any concern about fluid losses and refills. Further integration with the building structure and/or renewable energy sources may be explored.

NOMENCLATURE

c_P, c_V	Specific heats ($J \times kg^{-1} \times K^{-1}$)
k	c_p/c_v (-)
$\dot{m}$	mass flow rate ($kg \times s^{-1}$)
P	pressure (kPa)
T	temperature (K)
e	specific energy ($J \times kg^{-1}$)
U	global heat exchange coefficient ($W \times m^{-2} \times K^{-1}$)
$\dot{W}$	power (W)

β	compression / expansion ratio (-)
ε	heat exchanger effectiveness (-)
η_{vol}	polytropic efficiency (-)
λ_C, λ_E	polytropic exponent (-)
τ	time (s)
Subscripts	
amb	ambient
C	compression
E	expansion

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